

The impact of heat exchanger fouling on the optimum operation and maintenance of the Stirling engine

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Abstract

This paper focuses on the effect of heat exchanger fouling on the performance of the Stirling engine in combined heat and power (CHP) application. Fouling results from using biomass fuels and affects the heat exchanger that transfers heat into the engine. This heat exchanger is referred to as the heater. The heat exchanger that recovers heat from the flue gases is also affected by fouling. To determine the performance of the Stirling engine, a commercial Stirling analysis tool is applied together with models that have been developed for the heat transfer in the heater, regenerator and cooler of the engine. The Stirling engine model uses constant temperatures for the heat addition and rejection, with the theory of displacement engine as a basis. The fouling in the heat exchanger is taken into account by using a fouling factor that corresponds with the degradation in the total heat transfer coefficient. The Stirling engine model together with the model for heat exchanger fouling makes it possible to estimate the effect of fouling on the performance of the Stirling engine. A cost model is developed for the engine to translate changes in performance into economy in CHP operation. In the studied application, the Stirling engine is operated by the heat demand. Together with the selected control method, performance and cost models compose a tool for the simulation and optimization of the system. The use of the models to determine the optimal cleaning interval of the heat exchanger surfaces is considered.

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Keywords: Stirling engine; Combined heat and power; Fouling; Cost models; Operation and maintenance

1. Introduction

Distributed energy systems are becoming increasingly popular in supplying for local energy needs. One contributing factor for the popularity is the possibility to bring electricity generation close to the end-user. If heat is also required, cogeneration technologies can be implemented to gain high energy conversion efficiencies. The Stirling engine provides one option for combined heat and power (CHP) production especially under the 30 kWe scale. The engine uses external firing, which allows the use of solid fuels unlike with diesel engines, for instance. Compared to gaseous and liquid fuels, solid fuels have high ash con-

tents, which contributes to the increased slagging and fouling of the heat transfer surfaces. The objective of this study is to model the effect of fouling on the performance of the Stirling engine when using solid biomass fuels. In addition, consideration is given for the use of the models to determine the optimal cleaning interval in CHP operation. The work is a part of a larger research project that investigates the applicability of the biofuel-operated Stirling engine for small-scale distributed CHP systems.

2. Stirling engine model

In this paper, simple isothermal models are considered while modelling the α -type Stirling engine. The α -type engine contains one work piston and one displacer piston in separate cylinders. These cylinders are coupled together through three series-connected heat exchangers: the heater,

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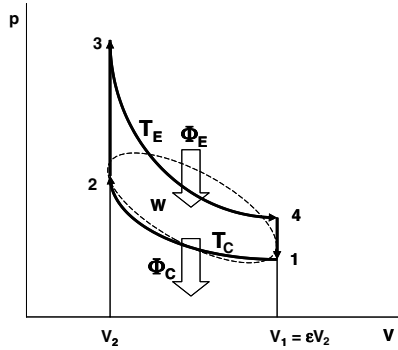


Fig. 1. The ideal Stirling cycle (continuous line) and the real cycle (dashed line) in the Stirling engine.

regenerator, and cooler. In the isothermal model, the heat from the flue gas is transferred to the cycle working gas during the isothermal expansion at a constant working gas temperature and at a mean pressure. An ideal Stirling cycle consists of the following sub processes (Fig. 1) [1]:

- The working gas is compressed to higher pressure (between points 1 and 2) by the work piston. The compression takes place at the constant lower temperature T_C . The cooler power Φ_C thus exits the cycle from the colder space. Heat is rejected and work is supplied.
- The displacer piston displaces the gas from the colder space to the hotter space. The gas is heated to the higher temperature T_E in the regenerator (between 2 and 3). The heating takes place at a constant volume since the work piston is not moving. No external heat is supplied to the system.
- The gas is expanded at the constant temperature T_E and thus the heater power Φ_E is imported to the cycle (between 3 and 4). Isothermal expansion takes place in the hotter space. The work piston moves down. Heat is supplied and work produced.
- The gas is displaced (by displacer) to the colder space and temperature T_C (between 4 and 1). The cooling process takes place in the regenerator at a constant volume. The work piston is not moving. The gas is cooled and heat is stored into the regenerator.

The area defined by the cycle on the pV diagram represents the work done. It is the difference of the supplied and rejected heats. The ratio of the work produced, W , to the heat supplied, Q_E , is defined as thermal efficiency η_{th} . In the ideal Stirling cycle, thermal efficiency depends only on the maximum and minimum temperature of the cycle, Eq. (1).

$$\eta_{th} = \frac{W}{Q_E} = \frac{Q_E - Q_C}{Q_E} = 1 - \frac{T_C}{T_E} \quad (1)$$

The above equation demonstrates the strong thermodynamic potential of the Stirling engine: the thermal efficiency of the ideal Stirling cycles equals to the Carnot efficiency, the highest possible value for a heat engine operating

between two temperature levels [2]. A real cycle is marked with the dashed line in Fig. 1.

The classical analysis of the operation of real Stirling engines is that of Schmidt [3]. The theory provides for harmonic motion of the reciprocating elements, but retains the major assumptions of isothermal compression and expansion and perfect regeneration. It thus remains highly idealized, but is certainly more realistic than the ideal Stirling cycle [4]. In a practical engine, aerodynamic flow losses in the heat exchangers cause a decrease in the area of the pV diagram and the net cycle output (indicated work). Analyses of the Schmidt type are of limited value, since the predictions must be modified by a multiplier in the range 0.3–0.4, in order to establish realistic values.

In this study, a commercial Stirling analysis program was adopted. The Snap2002 [5] computes the power output of the given configuration. The basic procedure follows Martini's [6] isothermal procedure. The code takes also into account the friction losses for the heater, regenerator and cooler, and estimates the following losses: mechanical friction loss, reheat loss (heat that must be added due to imperfect regenerator operation), shuttle conduction loss, static conduction losses, as well as pumping and temperature swing losses. The shuttling and pumping losses are hot cap or displacer gap losses. The temperature swing losses are due to the temperature of the regenerator matrix swinging when the gas is blown through it on each cycle.

3. Reference process

A reference process [7] was selected to test the modelling tools and to study the effect of fouling on a biomass Stirling engine. The specifications for the α -type engine (Fig. 2) are presented in Table 1.

The goal was to model the steady state operation of the Stirling engine inside the dashed line in Fig. 2, which is heated by the flue gas of a biomass furnace. The fuel is bark.

In Fig. 2, hot flue gas from the combustion chamber enters the heater of the Stirling engine and heat is transferred into the engine. In the engine, the heat is converted into work and lower temperature-level heat which is rejected in the cooler. The codes for the preliminary design of the heater, cooler and regenerator were written to give the right input values for the reference process configuration

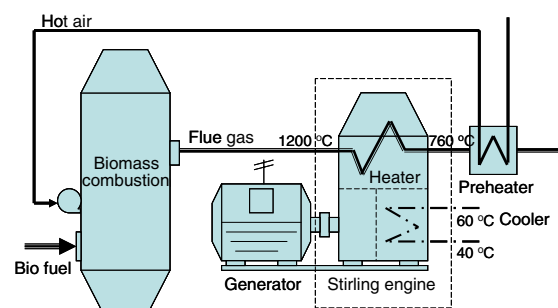


Fig. 2. Configuration of the reference engine.

Table 1
Specifications for the α -type reference engine

Average flue gas temperature	1000 °C
Thermal input (heater)	12.5 kW
Engine cooler	8.75 kW
Brake power	3.2 kW
Mean pressure	33 bar
Swept piston volume	840 cm ³
Working speed	600 rpm
Brake efficiency (brake power/thermal input)	0.25–0.28
Working gas	Air

modelled in the Snap2002 program. In the study, a tubular type heater and cooler and a screen type regenerator were used. The heater and the cooler were designed by the LMTD (log mean temperature difference) method and the regenerator by using the NTU (number of transfer units) approach [8,4,9,10]. In the heater and cooler, the expansion and compression take place at a constant temperature. Since the temperature change of one fluid is negligible, the heat exchanger behaviour is independent of the flow arrangement [8] or heat exchanger configuration.

The performance of the Stirling engine was calculated by the Snap2002 program using the thermal input Φ_E that is gained from the energy balances of the heater by iterative solution. The energy balances are based on the transferred heat from the flue gases, the log mean temperature difference method, and the ideal Stirling cycle assumptions on the working gas side. The following parameters were kept constant in the preliminary heater calculation: working speed and mean pressure of the Stirling engine, temperature of the flue gas before the heater, mass flow rate of the flue gas (18 g/s) and working gas (29 g/s), heat transfer coefficients, and thermal conductivity of the heater material. The compression ratio ε ($=V_1/V_2$) was 5 (Fig. 1). The calculations were performed for the steady state operation of the process using constant fuel power.

4. Fouling in the heat exchanger

According to Pålsson and Carlsten [11] solid bio fuels (e.g. wood powder) give high amount of ash residue when burned. Above a certain temperature the ash starts to melt and gets sticky, which leads to build-up of ash at the heater. It is imperative to keep the combustion chamber temperature below the melting point of the ash. Accordingly, the process simulations were carried out by assuming the flue gas inlet temperature of 1250 °C when using bark fuel (moisture 50%, ash 2.8%) [12].

Conventionally, fouling in the heat exchanger is modelled using total heat transfer coefficient U where the additional thermal resistances are presented as fouling factors [13]. For a tubular type, for instance, the total heat transfer coefficient is given by

$$U = \frac{1}{\frac{d_o}{\alpha_i d_i} + \frac{d_o}{2\lambda_w} \ln \frac{d_o}{d_i} + \frac{1}{\alpha_o} + \frac{d_o}{d_i} f_i + f_o} \quad (2)$$

In the equation, d_o and d_i are the outer and inner diameter of the tubes, α_o and α_i heat transfer coefficients on the corresponding surfaces, and λ_w thermal conductivity of the heater material, which is carbon steel in this study. f_o and f_i are the corresponding fouling factors. Tabulated values for these fouling factors are given for instance in [14]. In this study, fouling is considered only on the flue gas side of the heater.

Fouling affects the thermal input power to the heater. To study the effect of fouling on the performance of the Stirling engine, process parameters of the engine were calculated using different values for the fouling factor f_o . For preliminary evaluation, the fouling factor was estimated to vary from 0 to 40 m² K/kW. The values are comparable for the factors used in the case of Diesel exhaust gases, especially for finned tubes [14,15]. Actual f_o values for biofuel combustion will be gained by field tests of the test engine. For gas–gas heat exchangers, total heat transfer coefficients are typically in the range 6–35 W/m² K [16]. In this study, the values for the total heat transfer coefficient were between 12 and 48 W/m² K.

As the result of the simulations, Fig. 3 presents the effect of fouling factor f_o on the heater power Φ_E , cooler power Φ_C and brake power P_b . The effect on brake efficiency η_b (defined by the ratio of brake power to heater power) is presented in Fig. 4. The main input values for the simulations are given in Table 1.

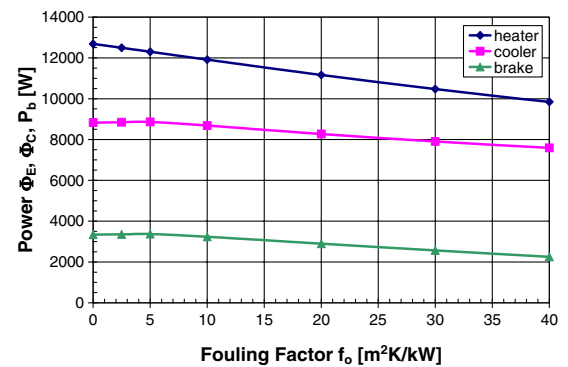


Fig. 3. Heater power Φ_E , cooler power Φ_C , and brake power P_b as a function of the fouling factor.

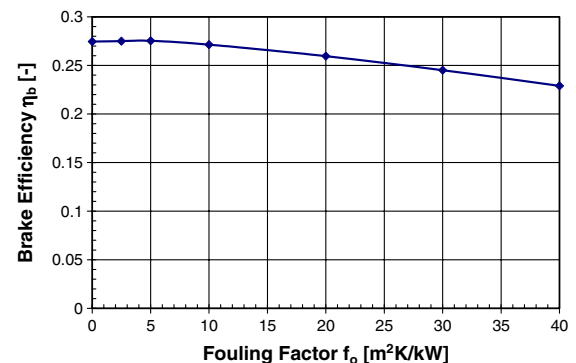


Fig. 4. Brake efficiency η_b as a function of the fouling factor.

It must be noted that the brake efficiency is determined as per heat input into the heater, not per fuel energy into the engine. Hence, the efficiency takes into account only the cycle losses and the mechanical losses in the real Stirling engine, but not the losses that prevail between the fuel power and the heater power. This makes the parameter less suitable for fouling studies. However, this type of efficiency is widely used in characterizing the Stirling engine performance, and is therefore used here.

As the figures show, fouling affects significantly the performance characteristics of the reference Stirling engine at the studied f_o range. When the heater is fouled, the heater power, cooler power, brake power and brake efficiency all decrease. The simulation results in Figs. 3 and 4 are comparable to the values of the reference process (Table 1) when the fouling factor is ca. 5–10 m² K/kW.

5. Combined heat and power production

In the combined heat and power application, heat is recovered from the flue gases that exit from the furnace. This is implemented in the heat recovery boiler. The recovered heat Φ_H can be utilized for water heating, for instance. In this study, the temperature of the flue gases is reduced only down to the level of 120 °C to safeguard against possible corrosion problems. The temperature of the water that enters the boiler T_{w1} is ca. 40 °C, while the exit temperature T_{w2} is ca. 60 °C. The same temperature levels prevail in the water-cooled Stirling engine cooler, which enables to use the heat Φ_C from the cooler for heating water as well. A schematic presentation for the studied Stirling engine in CHP operation is given in Fig. 5.

Based on the simulations performed for the reference process, Fig. 6 presents the effect of the heater fouling on the maximum amount of heat $\Phi_{H,max}$ that in theory can be recovered in the boiler. Fouling in the heater is indicated by the decrease in the heater power. As for recoverable heat, heater fouling has an adverse effect, governed by the First Law of Thermodynamics: the less heat is taken into the Stirling engine, the more remains to be recovered.

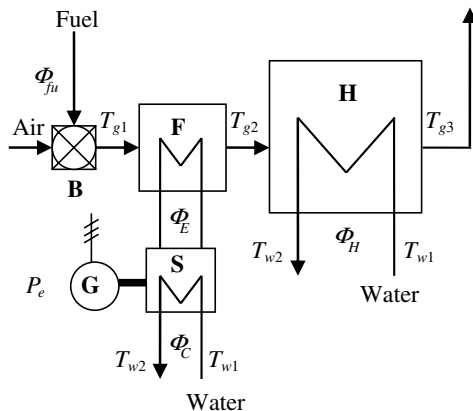


Fig. 5. The Stirling engine in CHP operation. B refers to the burner, F furnace, S Stirling engine, G generator, and H heat recovery boiler.

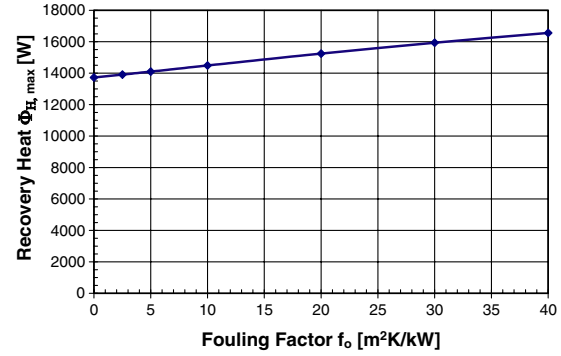


Fig. 6. Maximum amount of recoverable heat $\Phi_{H,max}$ as a function of the fouling factor of the Stirling engine heater.

To maintain structural simplicity in the configuration and hence, reasonable cost level, the studied engine has no air pre-heater unlike the reference engine. As a result, the electric efficiency of the engine suffers but the total efficiency of heat and power generation is favoured. The combustion air is assumed to enter the burner at reference temperature (15 °C) and the heat losses for the burner are assumed negligible. Hence, the fuel power Φ_{fu} into the burner can be determined from the steady-state energy balances written for the burner and heater.

$$\Phi_{fu} = \Phi_E \frac{h_{g1}}{h_{g1} - h_{g2}} \quad (3)$$

In the equation, h_{g1} and h_{g2} refer to the specific enthalpy of the flue gas before and after the Stirling engine heater.

Denoting the generator efficiency with η_{gen} , the electric power output P_e from the Stirling engine is given by

$$P_e = \eta_{gen} P_b \quad (4)$$

The electric efficiency η_e and the total efficiency η_{tot} of heat and power generation are defined using the net energy flows in the CHP system as follows:

$$\eta_e = \frac{P_e}{\Phi_{fu}} \quad (5)$$

$$\eta_{tot} = \frac{P_e + \Phi_H + \Phi_C}{\Phi_{fu}} \quad (6)$$

To summarize, Table 2 presents the main performance parameters for the studied Stirling engine configuration in CHP operation when no fouling occurs in the heater ($f_o = 0$). The value of 0.93 has been used for the generator efficiency in the calculation. The comparatively low value for the electric efficiency (10.1%) results from using moderate flue gas inlet temperature (1250 °C) and omitting the use of an air preheater.

In the studied CHP application, the engine is operated by the heat demand. Variation in the heat demand is indicated by changes in the mass flow rate of water into the boiler (Fig. 5). The temperature of the water T_{w2} that exits the boiler has a target value that the control system tries to maintain by varying the fuel flow into the burner. A proportional controller maintains the ratio of the air flow to

Table 2

Performance of the studied Stirling engine in CHP operation, no fouling in the heater

<i>Power (kW)</i>	
Fuel	30.8
Heater	12.7
Electric	3.11
Engine cooler	8.83
Heat recovery boiler	13.7
<i>Efficiency</i>	
Electric	0.101
Total	0.832

the fuel flow so that the flue gas temperature T_{g1} before the heater is constant at varying heat load levels. The temperature of the cycle working gas is kept constant in the heater by adjusting the amount of the gas in the Stirling engine.

Compared to the operation of a clean engine, fouling of the heater causes a decrease in the electric power output and electric efficiency, and increase in the flue gas temperature T_{g2} after the heater. Apart from the heater, the heat recovery boiler is also subjected to fouling. The effect of the increase in T_{g2} on the heat output Φ_H from the boiler and the water temperature T_{w2} exiting the boiler depends on the fouling rate in the boiler: at low enough fouling rates, Φ_H and T_{w2} both increase. At severe enough fouling, Φ_H and T_{w2} may also decrease. The control system tries to maintain T_{w2} , and hence the response from the control system to fouling is governed by the degree of fouling in the boiler.

6. Optimal operation and maintenance

As a result from fouling on the heat exchanger surfaces, the performance of the Stirling engine is affected, which in turn affects the on-line costs of operation. The fouling can be removed, but the removal induces cleaning costs. Additionally, the engine must be shut down for the cleaning. The unavailability during the cleaning is also a source for costs, sometimes of paramount importance.

The fouling and cleaning of the heat exchangers are a recurring phenomenon. Therefore, the optimal cleaning interval can be determined by adding up the aforementioned costs for one period that includes the time for the fouling to develop τ_f and the time that is required for the cleaning τ_c . The average cost flow at the period is gained by dividing the costs by the corresponding time interval. The optimization is based on determining the fouling time that yields the minimum cost flow. Cleaning time is assumed independent of the fouling time. The cleaning is performed for both heat exchangers at the same time.

Denoting the costs that are due to changes in engine performance with C_p , cleaning costs with C_c , and costs that are due to the unavailability during the cleaning with C_u , the average value for the total cost flow $\dot{C}_f$ from fouling at the period is given by

$$\dot{C}_f = \frac{1}{\tau_f + \tau_c} (C_p + C_c + C_u) \quad (7)$$

If the heat demand is constant, the most straightforward method to model the costs that result from changes in engine performance is using actual performance parameters: the fuel input and electric output. Since the engine is operated by the heat demand, the heat output is not affected by fouling. The effect of fouling on the performance depends not only on the amount of fouling but also on the level of the heat load. The impact of the load level can be eliminated by monitoring the averaged development of the fouling factors in the engine heater and in the heat recovery boiler. These fouling factors are denoted with $f_{oE}(t)$ and $f_{oH}(t)$. The fouling factors are then applied together with the developed Stirling engine models and the selected control method to evaluate the development of the performance parameters of the engine. Since the level of the heat load affects also the fouling speed, the optimization must be performed using the load level that corresponds with the average heat power $\bar{\Phi}_h$ at the selected monitoring period. The heat power Φ_h is defined as the sum of boiler power Φ_H and cooler power Φ_C . Thus, the following applies for the modelled fuel power Φ_{fu} and electric power P_e :

$$\Phi_{fu} = \Phi_{fu}(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) \quad (8)$$

$$P_e = P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) \quad (9)$$

The averaged development of the fouling factors is determined from the instantaneous values using curve fitting. The instantaneous values for the fouling factors are determined from the measured powers and temperatures by applying the models inversely.

The costs that are due to the unavailability during the cleaning period τ_c include the additional costs from acquiring heat and power from alternative sources. If the electric power from the engine is higher than the power demand of the system, the exceeding power is sold. In this case, the costs due to unavailability must also include the corresponding lost revenue. As a result, the formation of the cost term depends on the magnitude of the average electric power demand $\bar{P}_{e,d}$ at the selected monitoring period and the modelled electric power output for a clean engine.

For $\bar{P}_{e,d} \geq P_e(\bar{\Phi}_h, 0, 0)$

$$C_u = \left[\bar{\Phi}_h \frac{e_{fu}}{\eta_{alt}} - \Phi_{fu}(\bar{\Phi}_h, 0, 0)e_{fu} + P_e(\bar{\Phi}_h, 0, 0)e_{e,b} \right] \tau_c \quad (10)$$

For $\bar{P}_{e,d} < P_e(\bar{\Phi}_h, 0, 0)$

$$C_u = \left[\bar{\Phi}_h \frac{e_{fu}}{\eta_{alt}} - \Phi_{fu}(\bar{\Phi}_h, 0, 0)e_{fu} + \bar{P}_{e,d} \cdot e_{e,b} + [P_e(\bar{\Phi}_h, 0, 0) - \bar{P}_{e,d}]e_{e,s} \right] \tau_c \quad (11)$$

In the equations, $f_{oE} = f_{oH} = 0$ indicates clean engine, $e_{e,b}$ and $e_{e,s}$ are the energy prices for purchased and sold electricity, and e_{fu} is the energy price for the fuel. η_{alt} is the efficiency of the alternative heat production. In this case, heat is produced in the boiler with the boiler efficiency η_{alt} and using the same fuel as the Stirling engine.

The costs that are related to changes in the engine performance at the fouling period τ_f are given by

$$C_p = \int_0^{\tau_f} \{ [\Phi_{fu}(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) - \Phi_{fu}(\bar{\Phi}_h, 0, 0)] e_{fu} + \dot{C}_e \} dt \quad (12)$$

In the equation, the cost flow term $\dot{C}_e$ takes into account the changes in the electric power output. The formation of the term depends on the magnitude of the average electric power demand at the selected monitoring period, the modelled power output for a clean engine, as well as for a fouled engine.

$$\text{For } \bar{P}_{e,d} \geq P_e(\bar{\Phi}_h, 0, 0) \text{ and } \bar{P}_{e,d} \geq P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) \\ \dot{C}_e = [P_e(\bar{\Phi}_h, 0, 0) - P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t))] e_{e,b} \quad (13)$$

$$\text{For } P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) < \bar{P}_{e,d} < P_e(\bar{\Phi}_h, 0, 0) \\ \dot{C}_e = [P_e(\bar{\Phi}_h, 0, 0) - \bar{P}_{e,d}] e_{e,s} + [\bar{P}_{e,d} - P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t))] e_{e,b} \quad (14)$$

$$\text{For } P_e(\bar{\Phi}_h, 0, 0) < \bar{P}_{e,d} < P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) \\ \dot{C}_e = [P_e(\bar{\Phi}_h, 0, 0) - \bar{P}_{e,d}] e_{e,b} + [\bar{P}_{e,d} - P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t))] e_{e,s} \quad (15)$$

$$\text{For } \bar{P}_{e,d} \leq P_e(\bar{\Phi}_h, 0, 0) \text{ and } \bar{P}_{e,d} \leq P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) \\ \dot{C}_e = [P_e(\bar{\Phi}_h, 0, 0) - P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t))] e_{e,s} \quad (16)$$

The optimum cleaning interval is determined by setting $\frac{\partial}{\partial \tau_f} \dot{C}_f = 0$. As a result, an equation is gained from which the optimum fouling period τ_f can be determined using root-finding methods.

Assume that the changes in the engine performance can be approximated linearly so that the following applies:

$$\Phi_{fu}(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) - \Phi_{fu}(\bar{\Phi}_h, 0, 0) = a \cdot t \quad (17)$$

$$P_e(\bar{\Phi}_h, 0, 0) - P_e(\bar{\Phi}_h, f_{oE}(t), f_{oH}(t)) = b \cdot t \quad (18)$$

In this case, the optimum τ_f can be solved in closed form and becomes

$$\tau_f = \sqrt{\tau_c^2 + \frac{2(C_c + C_u)}{a \cdot e_{fu} + b \cdot \bar{e}_e}} - \tau_c \quad (19)$$

In the above equation, $\bar{e}_e$ is the average price of purchased and sold electricity, weighted by the change (absolute value) in purchased and sold energy due to fouling at the fouling period τ_f . The changes are determined using the average electric power demand $\bar{P}_{e,d}$ as a basis. $\bar{e}_e$ depends on the optimizing variable τ_f and thus, the optimization proceeds iteratively.

7. Conclusions

Heat exchanger fouling affects significantly the performance of the Stirling engine in CHP operation. For detailed analysis of the effect mechanisms, a steady-state performance model is required that takes into account

the fouling factors of the following two heat transfer surfaces in the combustion gas path: the heater for the Stirling engine and the boiler for water heating. Together with the control method of the system, the model can be applied to simulate engine performance at given loads and fouling factors. The model can also be used inversely to evaluate the fouling factors using measurements from the engine.

The development of the fouling factors over time can be utilized to determine the optimal cleaning interval of the heat transfer surfaces when the performance model is combined with the cost model that is developed for the system. Fouling speed depends on the load level and therefore, the optimum results are representative for the corresponding operating conditions that prevailed when determining the development of the fouling factors.

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